REPORT No. 94

THE EFFICIENCY OF SMALL BEARINGS IN INSTRUMENTS OF THE TYPE USED IN AIRCRAFT

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SUMMARY.

This investigation was undertaken by F. H. Norton, physicist at the research laboratory of the National Advisory Committee for Aeronautics, Langley Field, Va., to supplement the rather meager data available on the construction and mechanical properties of small bearings and pivots suitable for use in aeronautical instruments. The static and running friction, for thrust and radial loads, was determined for several conical pivots and for plain cylindrical journals and ball bearings. Also the static rocking friction was measured for several conical and ball bearings under a heavy load, especially to determine their suitability for use in an N. P. L. type wind tunnel balance. It is found that for a given small load the conical pivots give less friction than any other type, and their wearing qualities, when hardened, are excellent. When the load exceeds about 1,000 gms., ball bearings give less friction than pivots, and, of course, stand shocks and wear better. Very small ball bearings are unsatisfactory because the proportional accuracy of the balls and races is not as high as in the larger sizes. For rocking pivots under heavy loads it was found that a ball and socket bearing was superior to a pivot resting in a socket. Vibration greatly reduces the static friction of a pivot.

RUNNING CONICAL PIVOTS.

The pivots oil best if mounted on the revolving part, and on very small pivots it is best to cut a small groove above the point for an oil stop, as on the balance staffs of watches. Tool steel, such as used for making taps and reamers, is most suitable for pivots. The pivots are

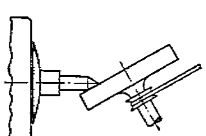
Fig. 1

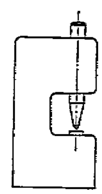
Fig. 2.

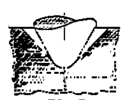
Fig. 3

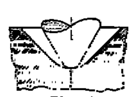
Fig. 4.


Fig. 5.

turned to size, hardened, and drawn very slightly. The point of the pivot can now be polished on a revolving lap, as shown in figure 1. Fine emery is used first, and then rouge, the hardness of the lap determining the radius of the point. A metal lap will give a small radius and a cloth lap a large one, but it is usually necessary to examine the point under a microscope and get the radius correct by hand lapping. It is not possible to grind a satisfactory pivot on a cylindrical grinder with an abrasive wheel, as the size of the grit, in even a fine wheel, approaches the diameter of the pivot at the point, and a very irregular surface is obtained, even though it may look smooth to the unaided eye. As only the extreme point of the pivot bears, this portion must be polished carefully; the surface on the remainder of the pivot is of little importance.

The sockets may be made by turning, turning and lapping, countersinking a small hole, or by punching, the last being by far the most satisfactory method. It is necessary to make a punch of the correct angle and radius of point in the same manner as the pivots, and it should be hardened and polished with the same care. It is convenient to hold this punch in a guide over the center of the blank socket, as shown in figure 2. It is very important that the punch be struck only one blow, which gives a socket as highly polished as the punch. If more than one blow is struck, the polish is lost. The socket is now hardened to the same degree as the pivot. It is important that the radius at the bottom of the socket be equal to that on the end of the pivot (fig. 3), for if it is larger the pivot will slide around (fig. 4), and if it is smaller (fig. 5) the friction is considerably increased. Before assembly both pivot and socket must be carefully cleaned to remove any chips or grit, and when together should be oiled with a light oil, such as watch oil, more for protection than for lubrication.

The running friction was determined by mounting a flywheel on the pivots to be tested, as shown in figure 6. The wheel was driven so that its speed was always the same at the start and then allowed to come to rest by its own friction, the time being taken, thus giving the relative running friction. The weight of the moving parts was 160 gms. and their moment of inertia 425 cm.^{2}, so that the radius of gyration was 1.63 cm. In order to obtain some idea of the actual value of the running friction, one set of pivots was tested with the air friction eliminated in the following manner: A hollow brass case was fitted around the wheel and belted, so that it could be driven at any speed. A small slit in the rim of the case allowed the wheel to be started by the friction of a small rubber disk mounted on a motor. This slit also allowed a black spot on the rim of the wheel to be observed. The wheel was first driven by the motor to a given speed, and the case was revolved in the same direction and speed as the wheel. It was possible to keep the box within a few revolutions a minute of the wheel at high speeds by the stroboscopic effect of the slit in the box and the spot on the wheel. At very low speeds it was more difficult to keep the speeds equal, but the air friction is almost negligible at this time.

As the only friction is pivot friction and the moment of inertia and initial speed of the wheel are known, the running friction may be found. Using the same wheel and pivot without the case, we have the same initial kinetic energy and pivot friction, but also air friction, which varies as the square of the rotational speed.

If T = average frictional moment of the pivot,

and A = average frictional moment of the air,

$$\text{K. E.} = \tfrac{1}{2} I\omega^2 = \tfrac{1}{2} T\omega t + \tfrac{1}{3} A\omega t$$

When $A = o$, t, the time to come to rest, is known for one case, and knowing I, the moment of inertia of the wheel and ω, its initial velocity, T, can be solved for. Now, substituting this value of T in the equation and calling t the time to come to rest in the open air with the same pivots, we can solve for A, the value of which is independent of the nature of the bearings used. Assuming that the friction is proportional to the time required to come to rest, it is possible by substituting in the equation, this time for any pivot, to obtain the approximate running friction for that pivot. The result obtained in this way can only be an approximation, as it involves the assumption that the frictional moment varies with velocity in accordance with the same law for all pivots, and also that the air frictional moment varies in the same manner. If it be assumed, for example, that the running friction varies with speed less rapidly than does the air friction (as is generally the case) the effect of the air friction will be less than that computed when the pivot friction is smaller than in the case used as a basis for the computations, and larger when the pivot friction exceeds that value.

The value of A was computed in one case and found to be 0.203 gm. cm. Since this moment acting alone would bring the wheel to rest in 13 minutes, and since the best pivot ran for over 20 minutes in free air it is evident that, as predicted in the preceding paragraph, the computed air friction is too high to be used in correcting the results with exceptionally good pivots. Tests were made with the shaft horizontal, and also with the shaft vertical, and the lower pivot acting as a step bearing.

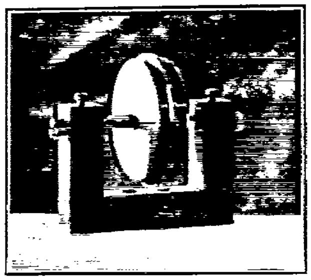

FIG. 6.

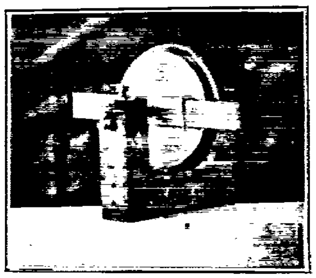

FIG. 7.

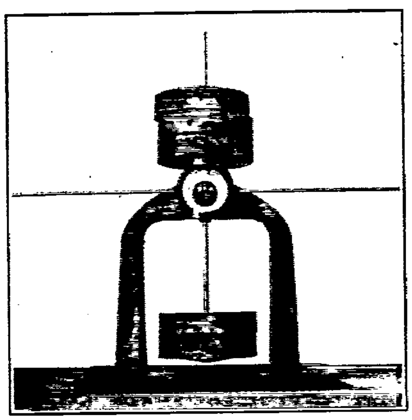

FIG. 13.

The static friction was determined by placing a graduated beam on the pivot axle and sliding a rider of known weight out from the center until the axle started to revolve. This point was found on each side of the axle and the mean of the two readings taken as the moment arm. It was found that the static friction varied for different positions of the axle, presumably because of minute irregularities in the pivots and sockets, so several readings were taken and averaged. The static friction was also determined for the shaft alone. The apparatus for doing this is shown in figure 7.

In the following table are given the properties of the conical pivots tested:

CONICAL PIVOTS.

Pivot.	Socket.	Time to come to rest from 3,000 r. p. m.		Starting moment.		Centering.	Wear.	Remarks.
		Vertical.	Horizontal.	160 gms. weight.	25 gms. weight.			
		′ ″	′ ″	*cm. gm.*	*cm. gm.*			
60°, fair polish, hardened, 0.001-inch radius.	75°, punched, 0.001-inch radius.	8 47	7 27	0.170	0.023	Very good		
Do	90°, punched, 0.001-inch radius.	11 45	9 00	.151	.011	Good		
Do	120°, punched, 0.001-inch radius.	12 22	9 22	.132	.011	Poor		
60°, rather rough soft steel, 0.002-inch radius.	90°, punched, 0.001-inch radius.	10 37	7 12	.289	.066	Good		
Do	90°, turned, 0.005-inch diameter flat spot at bottom.	8 21	8 12	.292	.033	Very poor		
Do	90°, turned, 0.013-inch hole in center.	2 50	4 10	.530	.132	Very good	Groove worn on spindle.	
Do	Coradi socket from planimeter, 90°, 0.004-inch hole.	9 55	6 50			Good	Groove worn on Coradi spindle.	Pivot broke during test.
60°, highly polished, hardened, 0.0005-inch radius.	90°, punched, hardened, high polish.	20 30	15 15	.080	.008	do		Polished on revolving lap; no wear; static friction inaccurate.
60°, highly polished soft steel, 0.0005-inch radius.	90°, punched, soft steel, high polish.	8 00	7 00			Poor		Wore badly.

Results are for two pivots, one on each side of wheel.

In figure 8 are plotted curves showing the effect of the socket angle on the friction. A 75° socket gives the best centering, and the 120° the least friction, but as a general thing a 90° socket will be found most satisfactory and is almost always used in instruments. Sockets

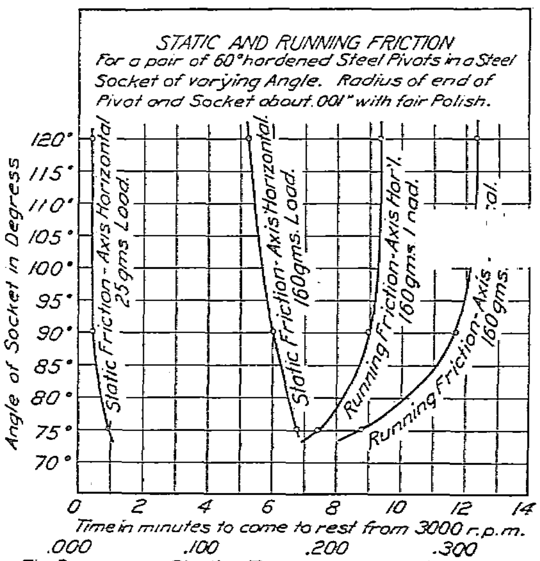

Fig. 8.

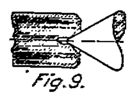
Fig. 9.

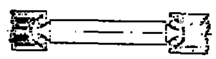
Fig. 12

with holes in the bottom give good centering but have considerable friction. An enlarged section of this type of socket, which is used on some planimeters and similar instruments, is shown in figure 9.

As shown in the preceding table, two similar sets of pivots and sockets were tested, one hardened and the other soft. The hard pivot and socket showed no signs of wear, while the soft ones wore so badly they rattled around loosely in the sockets and gave more than twice the friction of the hardened ones. Pivots were tried with and without oil and showed no appreciable difference in either static or running friction.

The radius at the point of the pivot can be made as small as 0.0005 inch when it is desired to reduce the friction to a minimum, giving a starting torque of only 0.0005 gm. cm. per gm. of weight. A pivot of this sharpness can not be used for loads much greater than 150 gms., and it is advisable for continuous running to make the radius twice as large as this. If the pivots and sockets are hardened and highly polished they make excellent bearings for light loads and give less friction than any other type.

METHODS OF HOLDING AND ADJUSTING PIVOTS AND SOCKETS.

Sockets must be capable of a fine adjustment and when adjusted be solidly supported. Several satisfactory methods of doing this are illustrated below. In figure 19 is shown the method used by Coradi. This is an excellent arrangement, as the socket is not rotated while

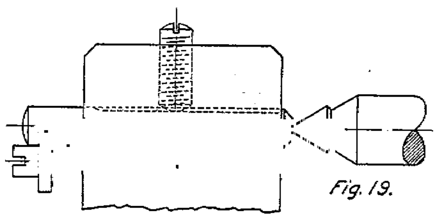

Fig. 19.

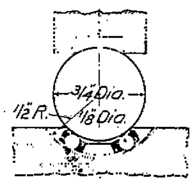

Fig. 17.

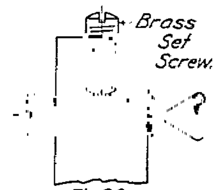

Fig. 20.

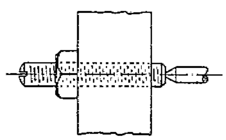

Fig. 21.

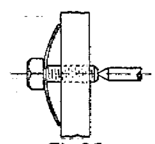

Fig. 22.

being adjusted and can be firmly clamped in place. It takes up considerable space, however, and can not be used in some locations for this reason. Another method is shown in figure 20 that is quite satisfactory and simple to construct. A screw and lock nut is sometimes used (fig. 21) but is not suited to fine adjustment. Figure 22 shows the method of locking the balance sockets in clocks by means of a cupped spring washer. This method gives only a small adjustment and rather insecure locking.

RUNNING CYLINDRICAL BEARINGS.

The cylindrical pivots were turned as smoothly as possible on a light lathe and were given a fair polish with crocus cloth, but were not hardened. The sockets were drilled with an ordinary twist drill and were not lapped out. The pivots were an easy fit in the sockets, but no looseness could be felt when they were oiled. The friction did not, however, seem to be altered by any reasonable amount of looseness. The static and running friction was determined in the same way as before, but the bearings were run in until the friction was constant.

All bearings were oiled with porpoise-jaw oil. The results obtained are given in the following table:

CYLINDRICAL BEARINGS.

Pivot.	Socket.	Time to come to rest from 3,000 r. p. m., shaft horizontal.	Starting moment— 160 gms. weight.	Starting moment— 25 gms. weight.	Wear.	Mean running friction.	Remarks.
		′ ″				*gm. cm.*	
1/64 [illegible]	[illegible]	0 44	4.70	0.60	None	2.9	
1/64 [illegible]	[illegible]	1 38	1.92	.24	do	1.27	
1/64 [illegible]	[illegible]	2 10	.89	.27	do	.85	
1/64 [illegible]	[illegible]	4 03	.53	.19	do	.37	
1/64 [illegible]	[illegible]	4 32	.50	.11	Considerable	.34	
1/64 [illegible]	[illegible]	4 32	.39	.03	do	.34	Pivot broke fourth run.
1/64 [illegible]	[illegible]	1 28					[illegible]
1/64 [illegible]	[illegible]	4 35					[illegible]
[illegible]	[illegible]		.08				[illegible] plane.
1/64 inch long, average diameter 0.0964 inch	do		.21				Do.

Results are for two bearings, one on each side of wheel.

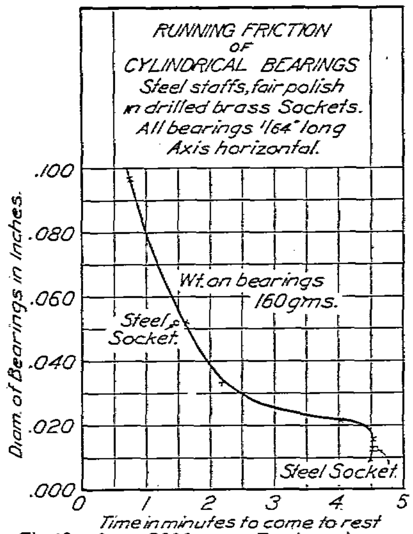

Fig. 10.

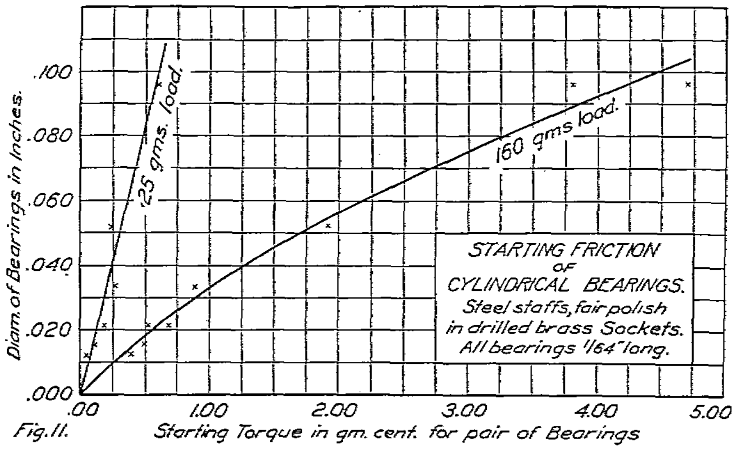

Fig. 11.

A curve of bearing diameter against running time is plotted in figure 10. The friction decreased rapidly with the diameter, until about 0.035 inch is reached, then the slope of the curve becomes much less down to a diameter of 0.020 inch. At this point the curve turns down sharply, so that no decrease in friction is obtained by reducing the diameter further. It is probable that the oil film breaks down at this point, as evidenced by the rapid wear below this diameter. For diameters in excess of this critical value the curve is approximately a rectangular hyperbola, indicating that the frictional moment is directly proportional to the diameter, and, therefore, that the frictional force acting at the periphery depends only on the magnitude of the total load and not on the intensity of pressure. It is evident that a small bearing should not carry over 500 pounds per square inch of projected area and 300 pounds would be safer. Two sizes of socket were tried of steel but the same result was obtained.

In figure 11 the static friction is plotted against bearing diameter for two loads. The values are rather irregular because of inequalities in the bearing surfaces, but it is evident that the curves are nearly straight lines starting at the origin and also that the ratio of the slopes of the two lines is the same as the ratio of the two weights. It may be concluded that the static friction increases as the weight on the bearing and as the bearing diameter, but is independent of the bearing length. The static friction of any small bearing (brass or steel, lubricated) is given by the formula following.

$$T = K\,D\,L,$$

where T = the starting torque in gm. cm.,
L = load in gms.,
K = 0.23 when D is in inches,
K = 0.0091 when D is in mm.,
D = diameter of bearing.

In order to determine the effect of lubrication two sizes of bearing were run with and without oil. It was impossible to detect any difference between oiled and dry bearings by the starting torque, but the oiled bearings had about half as much running friction as the dry ones. It is probable that the heavily loaded shaft cuts through the oil film when it is at rest, so that the same conditions of starting torque prevail whether oil is present or not.

The values of "mean running friction" given in the table were computed with due allowance for air friction in the manner already described. It will be noted that the mean lubricated running friction was only very little less than the static friction, and the mean dry running friction was therefore distinctly greater than the static. Since running friction at low speeds is always less than static friction, this excess must be attributable to variation of running friction with speed. It may be caused by heating, expansion, and partial seizure of the shaft, with a resultant great increase in friction when running at high speeds.

As the bearings of airplane instruments are used under conditions of vibration such conditions were simulated by placing the testing apparatus on a 2-horsepower electric motor frame and running the motor slightly out of balance at 1,800 r. p. m. In the case of a $\frac{3}{32}$ inch diameter bearing the static friction was reduced to less than one-twentieth of its steady value by vibration. The starting friction of a smaller pivot, 0.0215 inch in friction, is very marked, especially with large bearings, and should be taken into account when designing instruments for these conditions. The gain, however, is not quite as great as might at first appear, for the pivots must be made larger to stand the strains of vibrating conditions. Advantage is taken of this means of reducing friction in some sensitive wireless relays where a clock taps the frame of the instrument at short intervals.

In order to give a better idea of the size of bearings tested, the following table gives the diameters used in clocks and watches:

Type of staff.	Diameter.
	Inch.
Alarm clock escape and second hand	0.021
Alarm clock wheel up to minute hand	.030
Alarm clock main spring	.050
Watch escape and second hand	0.010-.013
Watch balance wheel, not jeweled	.007
Watch escape, jeweled	.007
Watch balance wheel, jeweled	.004

The cylindrical bearings have much more friction than conical pivots, but need not be hardened and do not require the delicate adjustment necessary with the pivots.

RUNNING BALL BEARINGS.

Three small ball bearings were tested for static and running friction. The first two were stock radial bearings, and the third was a cone bearing made with $\frac{1}{16}$-inch balls, as shown in figure 12. The description of the bearings and their friction is given in the following table:

Bearing.	Static friction—		Time to come to rest from 3,000 r. p. m.		Remarks.
	160 gms.	60 gms.	Horizontal.	Vertical.	
			′ ″	′ ″	
High-grade bearing: $\frac{3}{8}$ inch outside diameter, $\frac{5}{32}$ inch inside diameter, $8\frac{1}{8}$-inch balls; no retainer.	0.75	....	1 20	....	Ran smoothly.
Cheap pressed races: $\frac{11}{16}$ inch outside diameter, $\frac{3}{16}$ inch inside diameter, $15\frac{3}{32}$ inch balls; no retainer.	2.25	....	23	....	Races rough.
Cone and cup bearing: $\frac{7}{16}$ inch outside diameter, $\frac{1}{8}$ inch inside diameter, $10\frac{1}{16}$-inch balls; no retainer.	.27	....	3 15	3 30	

Friction is for a pair of bearings. All runs with dry bearings.

The first bearing was the most satisfactory and has about the same friction as a $\frac{1}{32}$-inch diameter cylindrical bearing and, of course, will carry enormously greater loads. It is evident that small ball bearings can not be made at present with enough accuracy to compete with pivots, under light loads (less than 1,000 gms.). However, for continuous running, and where it is necessary to pass the axle through the bearing, ball bearings can be used to advantage. The static and running friction is considerably increased by oil, but as this increase is independent of the load, oiling makes little difference on a heavily loaded bearing.

ROCKING PIVOTS.

The apparatus used to measure the static friction is shown in figure 13. It is simply a pendulum with horizontal arms to carry the riders. The weights are so placed that the center of gravity of the moving parts is at the pivot point, and one of the riders is pulled out until the pendulum starts to move. This is done for each rider in turn, and the mean reading is taken as the moment arm. In order to be sure that the pivot is unstrained at the beginning of the test, the end of one of the arms is held and the pendulum rocked slightly in a plane at right angles to the plane in which the test is carried out. As it was rather difficult to adjust the center of gravity exactly at the pivot point, the moment arm for the sharper pivots could not be accurately determined, and the values given may be in error as much as 50 per cent. In the first test the sockets were sharp at the bottom and had an angle of 120°, and the pivots were 90° with various radii at the points. In figure 14 is plotted the starting torque in gm. cm., against diameter of the point. With the lighter load the torque increases uniformly with the diameter of the point and the curve passes through the origin. As the load is increased the curves still start for the origin at large diameters, but as the diameters are decreased the curves fall below the straight line, due probably to flattening of the point after the unit load exceeds a certain value.

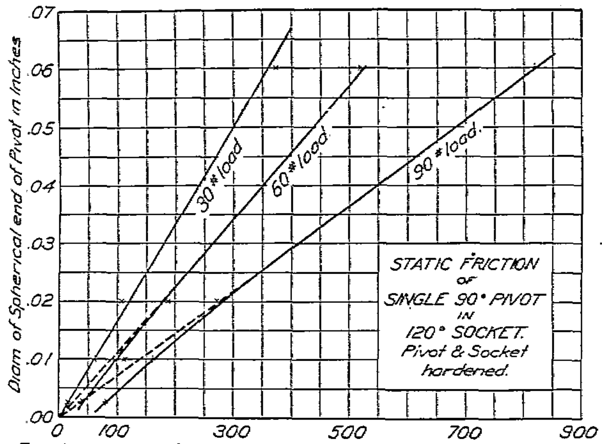

Fig. 14

Pivots that were loaded above this value showed wear and distortion. It seems evident that with a 120° socket and 90° pivot that the greatest permissible load will be given by:

$L = 3.000\ d$

where d is the diameter in inches of the pivot point

and L = total load in pounds.

The friction of these same pivots when resting on a flat plate, instead of on a socket, is about halved, but of course would stand very litte tangential load. In all cases when the loading was high the point left an indentation on the plate. One pivot and socket was inclined 15° in order to introduce a tangential component similar to that found in wind tunnel balances. The friction was the same as when the load line passed through the pivot axis.

As the same socket was used for all sizes of pivot, the rounded pivots rested in the socket as shown in figure 15, bearing on an annular area. If a socket had been used in each case to

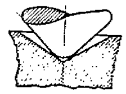

Fig. 15.

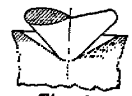

Fig. 16.

fit the pivot point the bearing would be as shown in figure 16, obtaining more area without increasing the diameter, thus carrying more load with the same friction, but with some danger of imperfect centering.

ROCKING BALL BEARING.

A hemispherical socket with a 1-inch radius was turned out, hardened, and lapped down with a steel ball as a lap to a good polish. A ring of $\frac{1}{8}$-inch balls held in a retainer was placed in the socket, and on these a $\frac{3}{4}$-inch steel ball rested. A section of the bearing is shown in figure

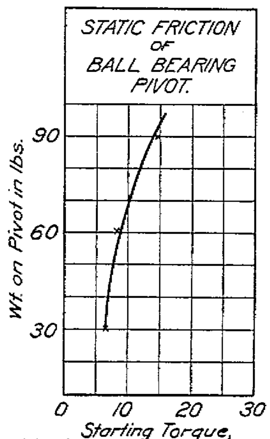

Fig. 18.

17. This bearing was tested in the same way as the pivots, and with all loads had less friction than the sharpest pivot, while it is evident that it could support many times the load (fig. 18). The small balls showed no tendency to crawl out of the socket, and the centering was very good. This type of bearing seems to be better than a pivot for wind tunnel balances.